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PLANETARY TRANSMISSION WITH A DISPLACEABLE COUPLING ELEMENT AND ACTUATOR

The invention is directed to a planetary transmission with a displaceable coupling element, by means of which the planetary transmission is shiftable, wherein the coupling element is displaceable by means of a shift fork that is movable by an actuator, and the actuator includes a motor and a cam that is driven by the motor via a shaft, and wherein the shift fork includes an element that engages a groove of the cam. The coupling element is generally a positive coupling, wherein the coupling teeth can be radially and also axially arranged. With a radial arrangement, the ring gear of the planetary transmission can also itself be a coupling element.

Such planetary transmissions, among others, are implemented in transfer cases of all-wheel-drive vehicles, in order to make available an on-road mode and an off-road mode.

A planetary transmission of this kind is known from EP 659 605 B1. With this planetary transmission, the cam roller is driven by the supporting shafts over a torsion spring. This serves as an energy accumulator, if the positive coupling element does not quickly locate itself in the coupled position. This construction is complex, the angular position of the cylindrical cam is never exactly known and no stop is available. Hence, the disconnection in the end position is also unreliable.

Furthermore, a planetary transmission is known from US 5,411,110, in which a rotatable disc that forms the shift cam cooperates with a sensing member, which is coupled to the coupling element via two springs. Here, the responsiveness of the interlock and the force distribution are depend-

ent on the difference of the forces of the two springs, which suffer from tolerances.

It is therefore the object of the invention to improve the shifting so that it is simpler, more reliable and more precise. It should establish a reproducible association between the angular position of the cam and the position of the shift fork and should yield upon exceeding a predetermined actuating force. In accordance with the invention this is achieved in that the cam is, in cross-section, a V-shaped groove with sloping sidewalls, and that the element of the shift fork is pressed by a spring into the groove. In this arrangement, the shift fork can be displaceable or pivotable and the cam can be disc or barrel shaped.

Through rotation of the cam, the rotational movement is transposed into a translational movement through the spring-biased element that engages the groove. This produces a precise guidance and positioning of the shift fork and additionally an overload safeguard. That is to say, if a specific guidance force is exceeded, the element climbs up one of the sloping sidewalls against the force of the spring. This happens when both coupling components stand tooth-to-tooth. The cam can then rotate further until in its end position. If the teeth are marginally misaligned then they are brought into engagement by the energy stored in the spring. In this arrangement, the target position is precisely defined again by the bottom of the groove. Still a further advantage is achieved in that: if shifting at low speeds, or if one of the coupling elements to be coupled experiences a speed increase, the reaction force acting on the electric motor is also limited.

In an advantageous and space-saving embodiment, the cam is essentially a cylindrical cam roller with a groove disposed on its surface (claim 2). In

this way, it is possible to provide the sidewalls of the groove, which is V-shaped in cross-section with independent, different angles of inclination (claim 3). In this manner, the threshold, at which the overload safeguard functions, can be provided differently for the two movement directions. In particular, a situation is achieved in that the inventive effect only occurs upon engagement of the coupling element, but not with disengagement. Hence, disengagement is also possible even when the coupling is not completely torque-free. Furthermore, the effect can be doubled without increase of the required packaging space, if two grooves are provided on the cam roller and an element of the shift fork engages in each of the two grooves (claim 4).

If, furthermore, the shift fork is not pivotably, in particular translatably, guided, a particularly attractive and compact embodiment is provided in which the shift fork has a tubular base surrounding the cam roller, which, together with the cam roller, forms a rectilinear guide for the shift (Claim 5). Consequently, the drive and the guide are combined by a pairing of components.

If, with this construction, the grooves are phase-shifted about a centering angle of 180° and the elements of the shift fork lie opposite to one another (Claim 6), the force exerted by springs of the elements balance one another. In this manner, the friction between the cam roller and the base is reduced. Higher precision, improved response of the interlock and low force requirement are the result.

In a compact and assembly friendly further development of the invention, the element of the shift fork is received within a spring containing cage, which is in turn mounted at a corresponding through hole of the tubular base (Claim 7).

The element of the shift fork is preferably a rotatably supported ball (Claim 8). This is not only kinematically ideal, it also reduces the friction and the demands on the performance of the electric motor. This in a particularly high measure, if the rotatable support of the ball is friction free.

In following, the invention will be described and explained with reference to the Figures. There is shown in:

- Fig. 1: a partial longitudinal section through a planetary transmission with an actuator in accordance with the invention,
- Fig. 2: a cross-section in accordance with AA in Fig. 1,
- Fig. 3: a variant of Fig. 2,
- Fig. 4: a cross-section in accordance with BB in Fig. 3, enlarged.

In Fig. 1, a planetary transmission is summarily indicated by 1, a central axis by 3 and an actuator summarily by 2. The planetary transmission comprises a primary shaft 4, a concentric secondary shaft 5 surrounding it with its sun gear 6, a planet carrier 10 with planetary gears 11 and a first set of clutch teeth 12, and finally a ring gear 15. One of the bearings 7 can be seen between the primary shaft and the secondary shaft and one of the bearings 8 can be seen between the secondary shaft and a housing 9, of which only a fragment is shown. The ring gear, shown in continuous line corresponding to a gear step of the planetary transmission, engages the clutch teeth 12 of the planetary carrier 10; in the other position, illustrated in phantom, the ring gear 15' engages in the second clutch teeth of the housing 9.

The ring gear 15 has a guide groove 18, in which the slide ring 21 of a shift fork 20 engages. The guide groove 18 is formed on the ring gear 15 in the illustrated exemplary embodiment. It can, however, also be located on any other shift movable component of the planetary transmission.

In Fig. 1 and Fig. 2 it can be seen, that the shift fork 20 has a tubular base 22, which surrounds a cam roller 23, upon which it is movable along the axis. The cam roller is rotatably fixedly connected to a shaft 26, which is rotatable in the bearings 24, 25 and is driven by a motor 27. The motor 27 is a controlled electric motor with or without a gear reduction drive. A groove 30 with a V-shaped cross-section can be seen at the periphery of the cam roller 23. The side walls 36, 37 (Fig. 1) of this groove are helical surfaces, which is indicated by the phantom line 31. The cross-sections of the groove 30 illustrated in Fig. 1 at the opposing generatrices are consequently profiles of one and the same groove.

An element that cooperates with the groove 30 is mounted on the tubular base 22. This element is here a ball 32, which is disposed in a piston 34 in a particularly low friction manner, the piston 34 being guided in a cage 33 and loaded by a spring 35. Thus, the ball 32 is pressed by the spring 35 into the groove 30 and in this way brings about the translation of the rotational movement of the shaft 26 to the displacement of the shift fork 20.

In the variant of Fig. 3 two balls 132, 132' in their cages 133, 133' are oppositely mounted at the tubular base of the shift fork 120. In Fig. 4 the 180° phase-shifted grooves 130, 130' are illustrated in enlarged form. It is also recognizable that the side walls 36, 37 of the groove 130' include angles 40, 41, which are different from one another, relative to the generatrix of the cam roller and with its central axis 42, respectively.

The operating mode of the spring loaded balls in cooperation with the grooves is the following: as long as the actuation force required for the translational movement of the ring gear 15 is normal, the V-formed grooves function as a groove with orthogonal walls, they produce a precise relationship between the angular position of the shaft 26 and the shift position of the ring gear 15, i.e., of the shift fork 20. If, however, a hindrance occurs upon displacement of the ring gear, for instance when the teeth of the ring gear 15 don't find their way into engagement with the clutch teeth 16 in the housing 9, then the ball 132 – as seen in Fig. 4 – climbs up the side wall against the force of the spring acting on it.

The actuation force at which this "overload coupling" begins to act depends on the pitch 40, 41 of the side walls and naturally from the force of the spring acting on the ball 132. When the described hindrance can only occur in one shift direction and not in the opposite direction, the angles 40, 41 can be selected to be different from one another. These angles also do not have to be constant over the entire length of the V-formed groove, they can also be variably designed in accordance with to the shift requirements.